

3004
226
JUN 12 1945

NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS

TECHNICAL MEMORANDUM

No. 1082

RECENT DEVELOPMENT OF THE TWO-STROKE ENGINE

II - DESIGN FEATURES

By J. Zeman

Zeitschrift des Vereines Deutscher Ingenieure
Vol. 87, No. 13/14, April 3, 1943

Reprint of R.T.P.3 Translation No. 2382
Issued by the Ministry of Aircraft Production,
London, England



Washington
May 1945

STANDARD DOCUMENT FILE



3 1176 01440 7754

NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS

TECHNICAL MEMORANDUM NO. 1082

RECENT DEVELOPMENT OF THE TWO-STROKE ENGINE

II - DESIGN FEATURES*

By J. Zeman

Completing the first paper dealing with charging methods and arrangements (VDI, vol. 87 (1943), no. 1/2, pp. 17-24), the present paper discusses the design forms of two-stroke engines. Features which largely influence piston running are:

- (a) The shape and surface condition of the sliding parts
- (b) The cylinder and piston materials
- (c) Heat conditions in the piston, and lubrication

There is little essential difference between four-stroke and two-stroke engines with ordinary pistons. In large engines, for example, are always found separately cast or welded frames in which the stresses are taken up by tie rods. Twin piston and timing piston engines often differ from this design. Examples can be found in many engines of German or foreign make. Their methods of operation will be dealt with in the third part of the present paper, which also includes the bibliography.

The development of two-stroke engine design is, of course, mainly concerned with such features as are inherently difficult to master; that is, the piston barrel and the design of the gudgeon pin bearing. Designers of four-stroke engines nowadays experience approximately the same difficulties, since heat stresses have increased to the point of influencing conditions in the piston barrel.

Features which notably affect this are:

- (a) The material
- (b) Prevailing heat conditions

*Z.VDI, vol. 87, no. 13/14, April 3, 1943, pp. 177-182.

NOTE: Reprint of R.T.P.3 Translation No. 2382; issued by the Ministry of Aircraft Production, London, England.

- (c) The shape, surface condition, and lubrication of the sliding parts

SHAPE AND SURFACE CONDITION OF THE SLIDING PARTS

The placing of the scavenging and exhaust ports affects the heat distribution in the cylinder and consequently constitutes a measure of the possible deformation in running. From this point of view, scavenging devices which do not cause distortion of the cylinder are to be recommended. A logical requirement is, of course, that the temperature is kept as uniform as possible across the cylinder. Alternatively, zones of high temperature on one cylinder side must be compensated by zones of low temperature on the same side.

It is easy to understand that all past experience is now being applied to perfecting the working surfaces. Chromium plating of the cylinder liners is apparently also coming into use. (See reference 104.)

CYLINDER AND PISTON MATERIALS

The most frequent material for cylinders (or liners) is cast iron, although steel is also often employed. The materials used for pistons are cast iron, steel, and light alloys. In cast iron and steel pistons is noted an increasing use of scraper rings (more rarely slipper guides) which are rolled or hammered in. They are made of white metal or more frequently of lead-bronze. (See figs. 33 to 36.) The pistons are then no longer stressed along their entire length, and possible cylinder distortion is consequently prevented.

HEAT CONDITIONS IN THE PISTON

Heat conditions in the piston are primarily influenced by the scavenging process. The flow of the exhaust gases often causes an increase in temperature on one side of the piston, and the scavenge air flow a temperature drop on the other. Piston crowns of high tensile strength are consequently required. (See figs. 33 and 34.)

Transmission of the heat accumulating in the piston crown

to the coolant (cooling water or oil) is a problem which designers are at present trying to solve in two totally different directions.

In the original and older principle the heat is conducted to the piston rings and the heat flow to them aided by liberal cross sections. (See figs. 33 and 34.) If the thermal loading is not too high, this method is adequate. In highly stressed engines, however, and also in opposed- and timing-piston engines where the pistons are on the exhaust side, the problem becomes considerably more difficult since the piston rings naturally tend to stick.

As in four-stroke engines, more recent designs, however, appear to follow an alternative principle, by which the major part of the abstracted heat is kept away from the piston rings so as to maintain their temperature as low as possible. To achieve this, the heat flow to the rings is therefore impeded by suitably reducing the ring cross section or providing cast-on carrier rings. The principle is thus, to conduct the heat accumulating in the piston crown either directly to the circulating oil or indirectly through the piston skirt to the cylinder and circulating oil, bypassing the rings. This implies a much simpler method of piston cooling, which is clearly illustrated in figure 37.

Another method consists of relieving the heat stresses on the piston rings by preceding packing rings designed to prevent seizing. The so-called "gas seal" (fig. 30) is the most common example of this, although other types have also been suggested. (See fig. 39.)

Finally, shields of heat-resisting steel are used (fig. 38). They can support higher working temperatures, so that the drop in temperature between the gas and the piston, and consequently the amount of heat to be abstracted from the piston are reduced.

Large or heavily heat-loaded pistons, as well as the pistons of all opposed piston engines must be completely cooled. (See figs. 35 and 36.) The coolant used is usually oil. Fresh water is used more seldom while air cooling has also been suggested (fig. 39). The coolant is conducted through jointed or telescopic pipes. Different methods of adapting the connecting rod for this purpose are being studied.

The piston gudgeon pins run without alternation of pressure. Consequently, it is particularly difficult to feed the lubricant to the sliding surfaces at high bearing pressures. This diffi-

culty can be overcome in large engines by reducing the surface pressure through an increase in the bearing dimensions. (See figs. 35 and 36.) Smaller engines usually have needle bearings. (See fig 37, also figs. 46 and 47.) In light alloy piston floating gudgeon pins are used. (See figs. 41 to 43.)

The other components of the power plant are developed similarly to those in other types of engines with the exception of the number of cylinders and order of firing, which features must be adapted to the two-stroke cycle. In regard to torque development, mass balance, and shaft oscillation, an odd number of cylinders is more favorable in most cases. However, in addition to eight-cylinder engines, satisfactory more recent arrangements are known in which an even number of cylinders is used.

LUBRICATION

In comparison with four-stroke engines the lubrication of two-stroke types presents greater difficulties. Since the exhaust ports tend to cause loss of oil, two lubricating systems are usually provided (in large engines always). They are:

- (a) Pressure lubrication, in which the quantity of fresh oil for piston lubrication is controllable
- (b) Forced lubrication for the remaining part of the power plant

In small types increased oil consumption is often considered inevitable, the pistons also being included in the forced-lubrication system. In engines with exhaust valves, oil losses do not exceed those in four-stroke engines, so that the same arrangements suffice.

In crankcase-scavenging engines all moving parts situated in the crankcase are necessarily lubricated with fresh oil. In Diesel engines the crankcases are therefore included in the forced lubrication system. In recent engines, the provision of means for automatic return of oil from the crankcase sump into the lubricating circuit (that is, resort to closed-circuit lubrication) has enabled oil consumption to be reduced to the level of that in any other engine. (See figs. 46 and 47.)

In crankcase-scavenging engines on the Otto cycle a mixture lubricating system, in which a fuel-oil mixture (usually

in 20:1 ratio) is introduced into the carburetor, is commonly used. The oil collecting in the crankcase is used for lubricating the transmission mechanism. This is a very effective method, but the oil consumption is necessarily high. It is, however, detrimental to export prospects, such fuel-oil mixtures usually being unobtainable at foreign filling stations. Fuel injection and the substituted gaseous fuels require different types of lubrication. Some of these appear promising (fig. 57, reference 68) and will be discussed in part 3 of the present paper.

GENERAL LAYOUT OF THE TWO-STROKE ENGINE

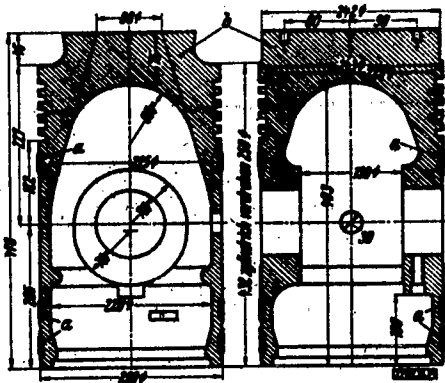
In general design engines with simple pistons are similar to other types. The practice of using separate cast or welded frames, in which the stress is taken up by tie rods, is always found in large engines (fig. 401) and occasionally in smaller types. In two-stroke engines, particularly, this form of construction is capable of fullest and most effective development, and results in pleasing and effective designs. Smaller and medium-size engines are usually fitted with continuous cast cylinder blocks extending to the cylinder tops, placed on a bed plate (references 77, 80, 99, 100, 102, 104, 105, 107, 109, 110, 111, and 113). The stresses are usually taken up by tie rods while the cylinder liners are separately fitted. This type of engine represents a stepping stone to the specifically automobile engine (figs. 41 to 45) of in-line or V-type, which is essentially similar to the four-stroke engine (references 74 to 76, 78, 79, 81 to 83, 103, and 112). Tie rods are often omitted in this case, however, the bed plate being formed by the oil sump and separate bearing caps.

Small and medium types of crankcase-scavenged Diesel engines have crankcases fitted with separate cylinders, the crankcase stresses being taken up by tie rods. (See figs. 46 and 47.)

The opposed-piston and timing-piston engines (references 85 to 98), on the other hand, often show considerable deviation from this standard design since their entire construction must, of course, be adapted to the special conditions. The Doxford engines shown in figures 48 to 49, the most popular type of large opposed-piston Diesel engine faithfully adhere to the Junkers system of construction (references 91, 92 and 95), while the Junkers engine itself, which is now-a-days built only in

smaller sizes, tends to develop along more economical and compact lines. In opposed- and timing-piston engines are further found all possible variants, of which the figures 50 to 56 give an approximate idea. The general impression is of an uncertain and transitional development. Only in small, high-speed spark-ignition engines (e.g., figs. 55 and 56) has the twin-parallel piston arrangement been standardized, partly perhaps, because the general layout need differ only slightly from that of standard-piston engines.

Translation by J. Helledoren



Figures 33 and 34.- Cast iron piston of a marine Diesel engine. (See figures 46 and 47)

- a, Hammered-in lead-bronze alloy guide rings.
- b, Thick piston crown.

Figures 35 and 36.- Cooled piston of an opposed piston engine. Type M.A.N.

- a, Floating gudgeon pin with increased bearing area.
- b, Lead-bronze alloy guide rings.

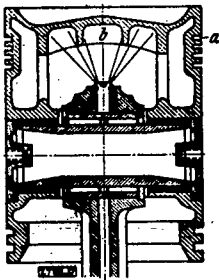
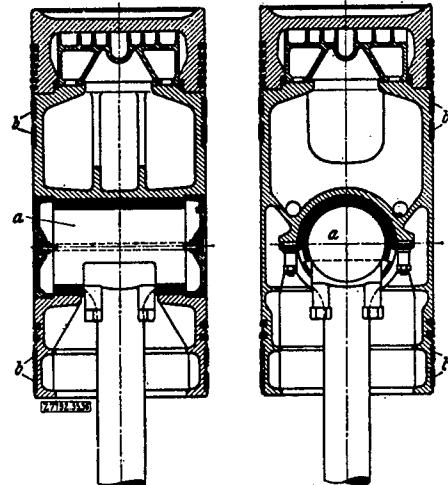


Figure 37.- Cast iron piston of a U.S.A. Diesel truck engine. (See figures 44 and 45). The heat flow to the rings (a) is interrupted by a reduction in cross section. The heat in the piston crown (b) is directly dispersed by splash oil (closed circuit lubrication).

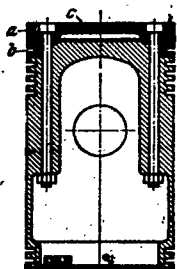


Figure 38.- Basic design of piston with gas seal ring. The gas seal ring (a) relieves the piston rings (b). The shield (c) made of heat resisting material reduces the quantity of heat transmitted to the piston crown.

Figure 39.- Sketch of a Woydt trapezoidal piston ring. The floating piston crown (a) is forced against the conical surface of the ring (b), and thus produces the sealing pressure.

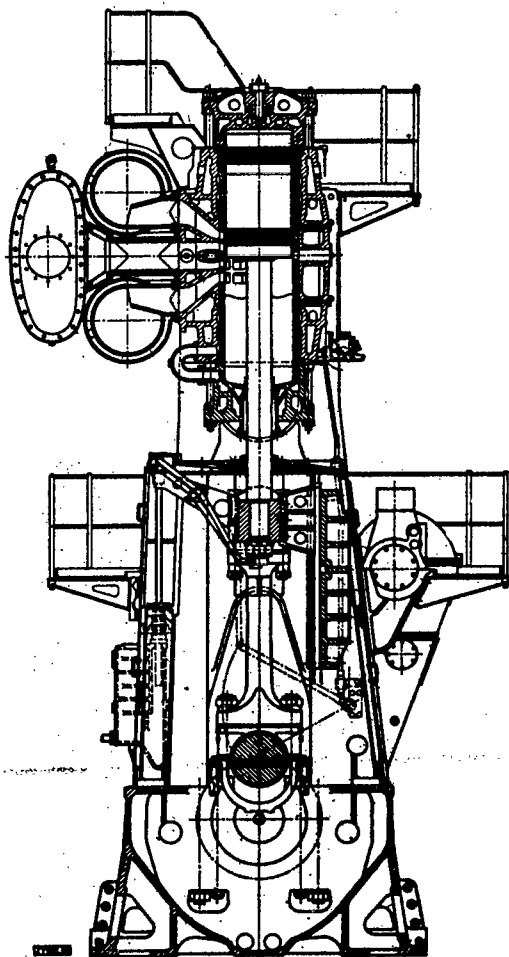
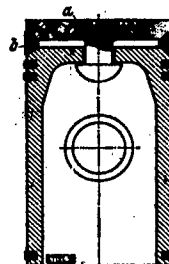
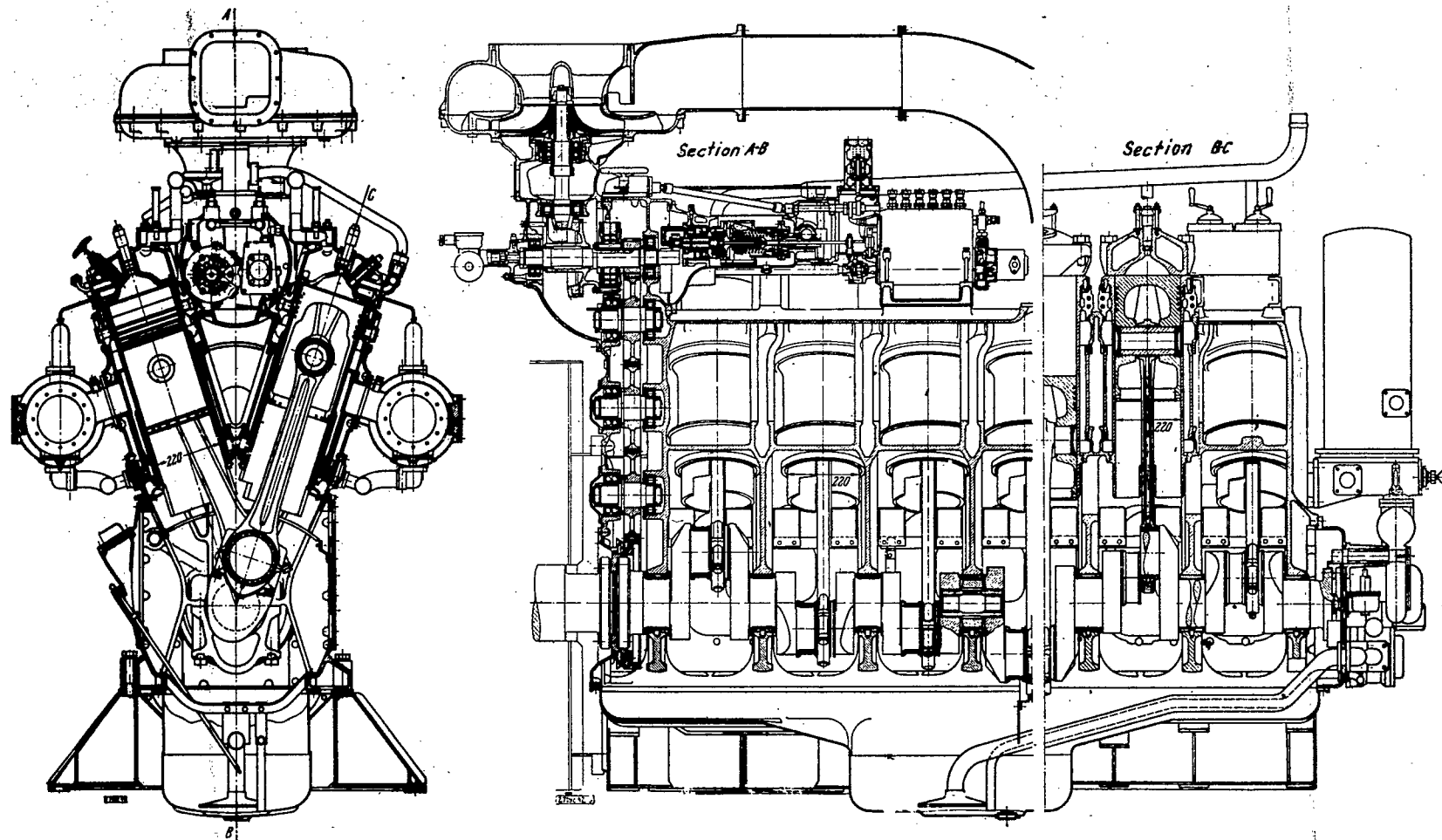
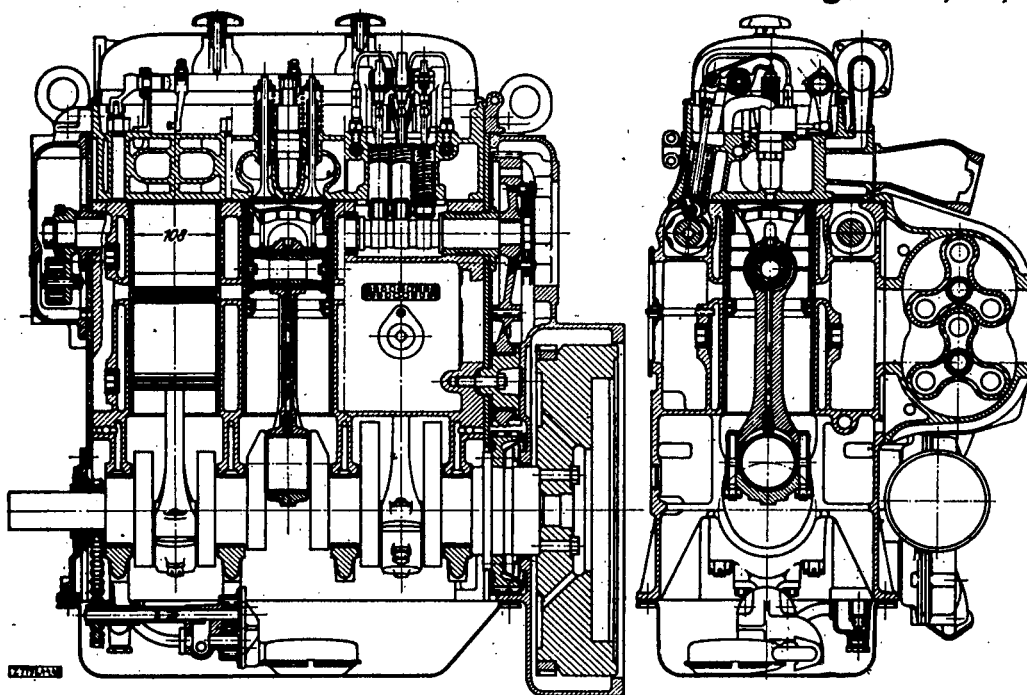


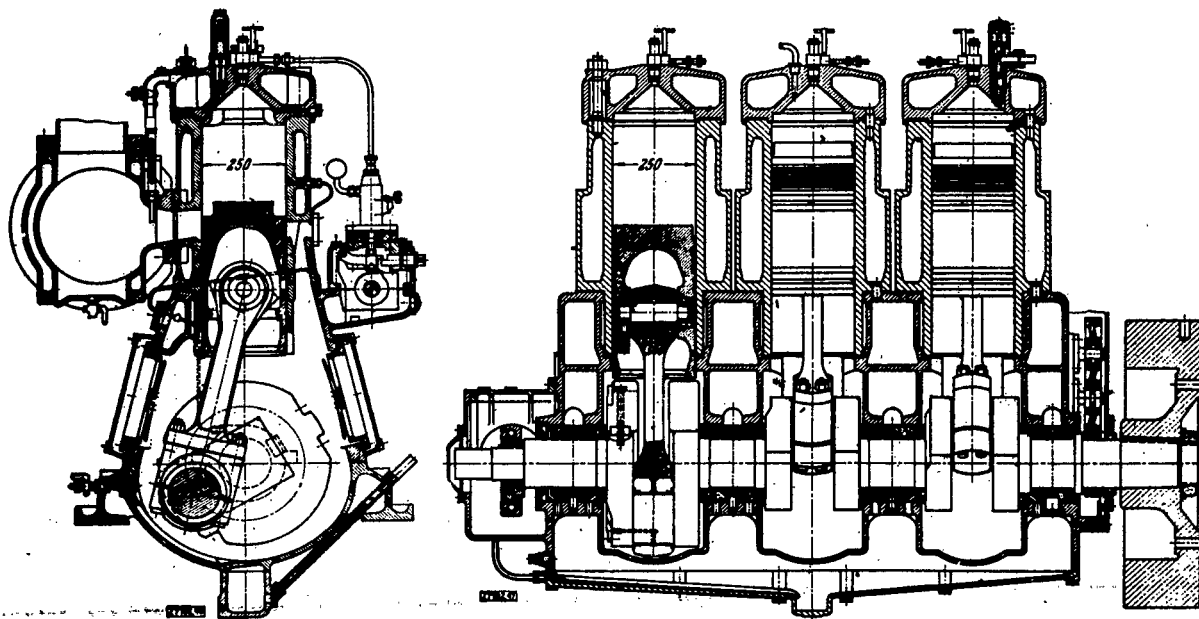
Figure 40.- Opposed piston type of Diesel engine with swirl scavenging. 800 hp, 140 rpm, M.A.N. Typical example of recent German large engine with separately cast frame. Stresses are taken up by tie rods. Comparison with figures 48, 49 and 50, 51 clearly shows the different approach to the design problem.



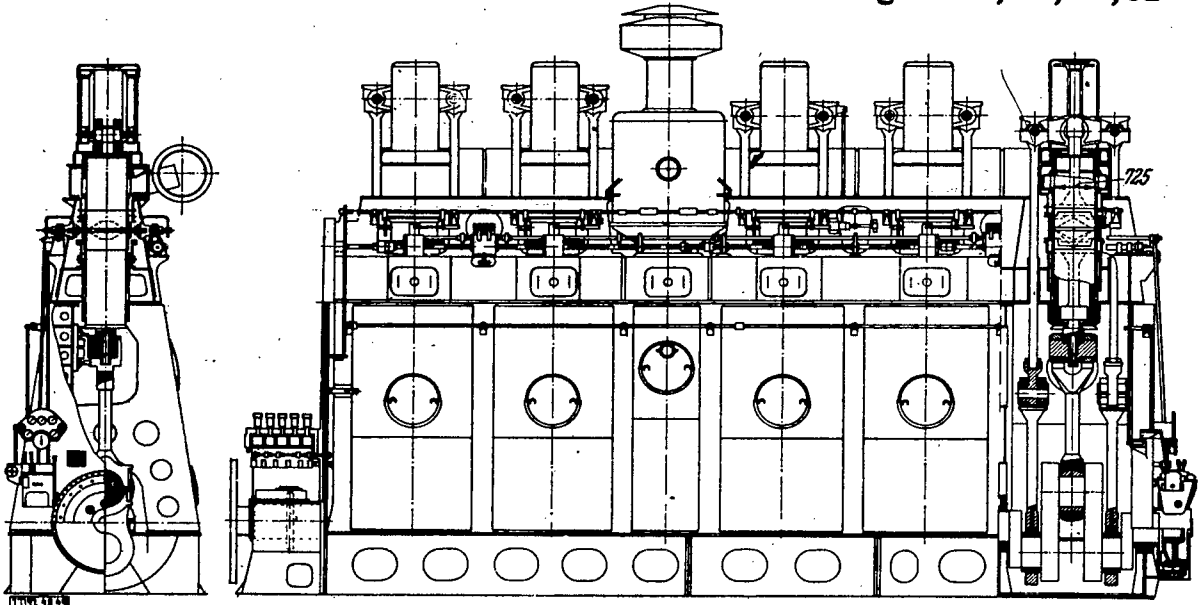
Figures 41 to 43.- Twelve-cylinder V engine, 1200 hp, 700 rpm, Klockner-Humboldt-Deutz type. 220 mm bore, 330 mm stroke, mechanically driven centrifugal blower, swirl scavenging, fuel injection. Latest German marine and automobile engine. For comparison see corresponding U.S.A. engine in figures 52, 53.



Figures 44 and 45.- Diesel engine with 37.5 hp per cylinder at 3000 rpm. General Motors Co. 100 mm bore, 127 mm stroke. Built as three, four and six-cylinder in-line engine. Uniflow scavenging, exhaust valves, mechanically driven Roots scavenging blower with helical blades.

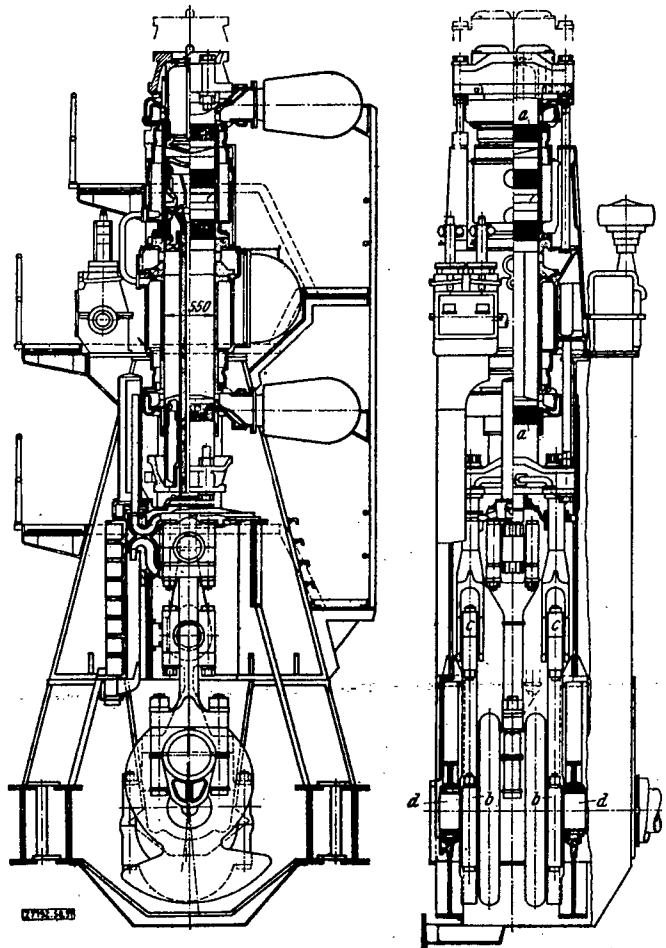


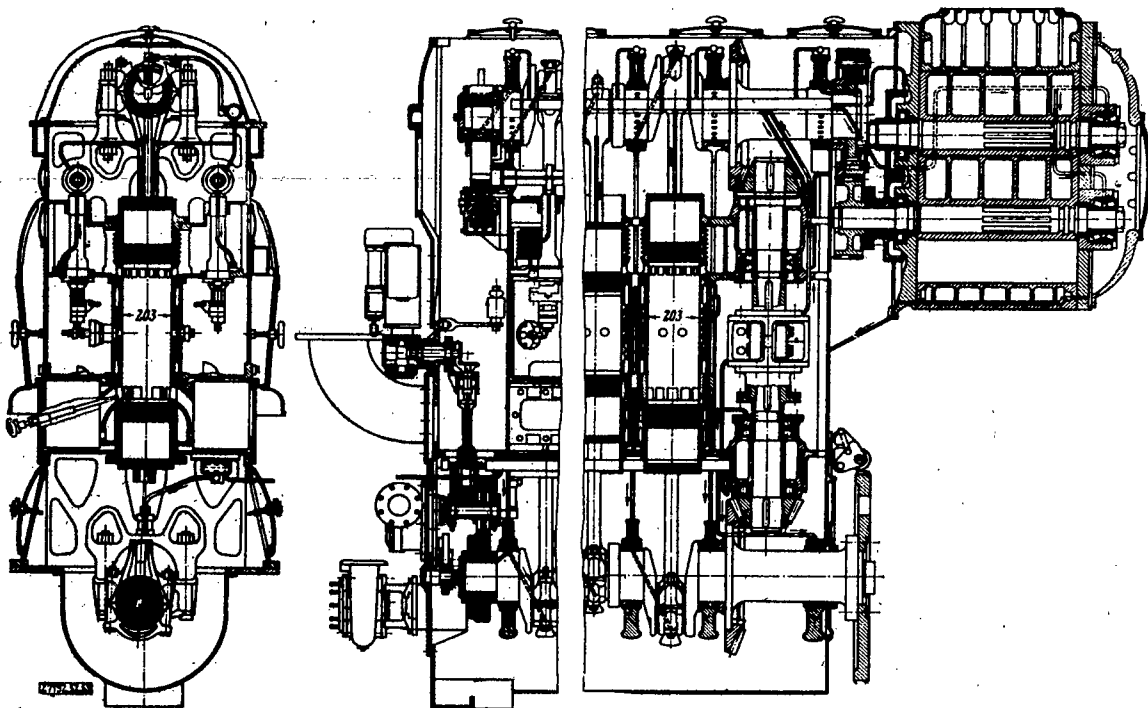
Figures 46 and 47.- Marine Diesel engine of 50 hp per cylinder at 325 rpm. Hanseatische Motoren Gesellschaft, Hamburg-Bergedorf. 250 mm bore, 350 mm stroke. Improved German crankcase scavenging. Separate cylinders. Crankcase stresses taken up by tie rods. A mep of 4 atm is obtained with satisfactory fuel consumption (177 to 175 g₃/hp) and low oil consumption little over 2 gr).



Figures 48 and 49.- 5-cylinder opposed piston engine. 8000 hp, 133 rpm, 725 mm bore, 1300 + 950 mm stroke. A popular British marine engine which has recently also gone into production in U.S.A. The original Junkers design has been retained practically without change.

Figures 50 and 51.- Latest design of double-acting engine with timing piston. 1100 hp per cylinder at 125 rpm. Burmeister and Wain type. 550 mm bore, 1200 mm piston stroke, 400 mm timing piston stroke. The timing pistons (a) are directly driven from the crankshaft (d) by the eccentrics (b) and gear train (c). Diameter of timing piston equals diameter of working piston.





Figures 52 and 53.- 800 hp opposed piston engine, 720 rpm. Fairbanks-Morse type. Mechanically driven Root centrifugal blower. 203 mm bore, 254+254 mm stroke. Rail car and marine engine. Copying the Junkers aero engine.

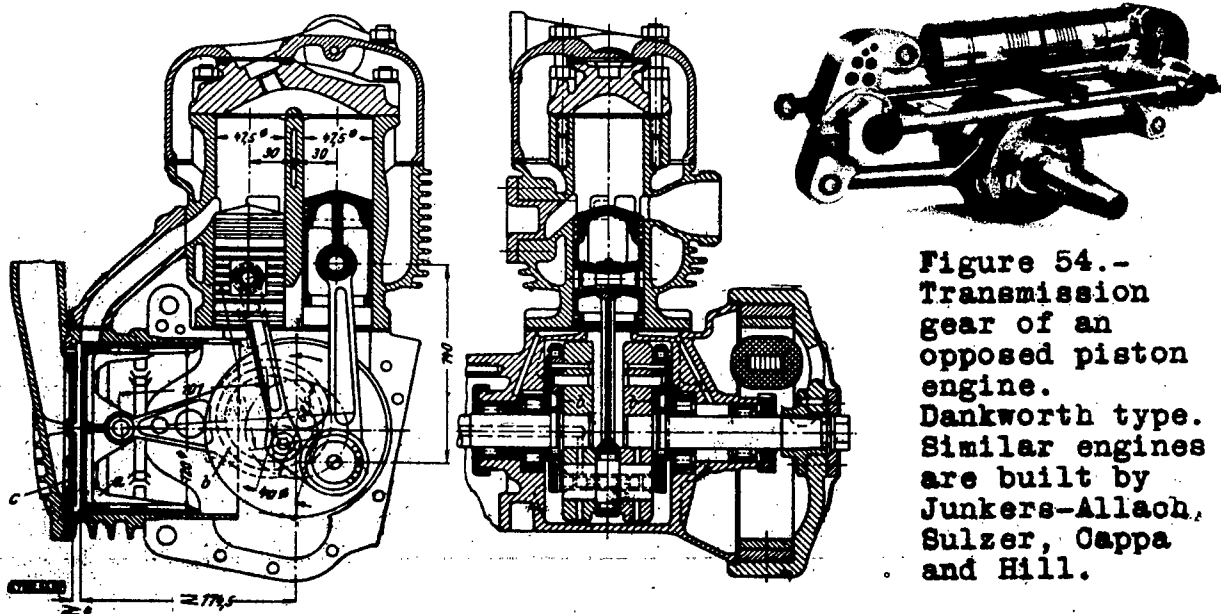


Figure 54.- Transmission gear of an opposed piston engine. Dankworth type. Similar engines are built by Junkers-Allach, Sulzer, Cappa and Hill.

Figures 55 and 56.- Two-stroke racing engine. UR 250 type of Auto-Union dated 1938. Separate reciprocating blower, of which the piston (a) is driven by the crankshaft through the eccentrics (b). Inlet ports controlled by louvres (c), twin-piston type.

NASA Technical Library



3 1176 01440 7754